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# Improving the performance of a Seawater Greenhouse desalination system by assessment of simulation models for different condensers

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ABSTRACT

The main aim of this paper was the development of a mathematical model for a new proposed passive condenser in order to enhance the performance of a humidification–dehumidification Seawater Greenhouse desalination system. Seawater Greenhouse desalination is used to create a cool environment and at the same time to produce fresh water for irrigation of crops grown inside the unit. The condenser in particular is currently one of the main bottlenecks in the commercialization of the technology. In addition to the current pump driven condenser, two new designs were considered: a passive cooling system with a condenser immersed in a water basin, and an external passive condenser connected to a basin of water placed on top of the cooling unit. The simulated condensate values for the proposed passive cooling condenser were compared with that of the actual measured values of the installed condenser. Preliminary results suggest that the passive condenser has a much greater water production capacity than the existing pump driven system. While the model for the proposed system still needs to be validated experimentally the initial study indicates that the passive containment cooling system is a promising improvement in the further development of greenhouse desalination.


**Contents**



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**Nomenclature**

| | |
|---|---|
| $C_p$ | heat capacity of the gas mixture |
| $D$ | diffusive coefficient |
| $D_t$ | tube diameter |
| $f$ | friction factor |
| $h$ | heat transfer coefficient by convection |
| $h_D$ | mass transfer coefficient |
| $K$ | conductivity |
| $L$ | tube length |
| $m''$ | flux of fresh water produced per unit of area |
| Pr | Prandtl number |
| Re | Reynolds number |
| Sc | Schmidt number |
| $T$ | temperature |
| $U$ | gas vertical velocity |
| $\nu$ | kinematic viscosity |

*Greek letters*

| | |
|---|---|
| $\lambda_m$ | thermal conductivity of the gas mixture |
| $\rho$ | gas mixture density |
| $\mu$ | gas mixture dynamic viscosity |

*Subscripts*

| | |
|---|---|
| w | inside wall |
| $\infty$ | bulk gas mixture |

## 1. Introduction

In many countries that suffer a chronic shortage of water, such as those of the Middle East and North Africa (i.e., MENA), over 80% of all fresh water consumed is used for agriculture. As fresh water resources are finite, there is an inexorable pressure to reduce agricultural use of water [1–7]. The Seawater Greenhouse, employing humidification–dehumidification processes, provides a possible solution to this dilemma by creating a growing environment that substantially reduces the amount of water required for irrigation, in addition to providing a new source of fresh water [2–6,8]. There is also a growing realization that the long-term solution to a shortage of potable water lies in a coordinated approach involving water management, purification, and conservation [1].

The humidification–dehumidification process (HD) is a versatile technique that can be adapted for water desalination [2,3,8] (Fig. 1). This method has several advantages such as flexibility in capacity, moderate installation and operating costs, simplicity, possibility of using low temperature and the use of renewable energy (e.g. solar, geothermal, recovered energy or cogeneration). Although the common methods of desalination such as distillation and reverse osmosis have been the subject of many investigations, studies of the HD process have been limited. A rigorous mathematical model describing the basic phenomena occurring in the greenhouse is required to design and evaluate the performances of new and existing systems. Other problems currently not addressed include the ineffectiveness of the greenhouse in high humidity environments as evaporating cooling is not possible or highly ineffective for certain types of plants. We can speculate that a possible solution could be the use of a dehumidifier prior to the first evaporator.

By its geographical location in an arid and semi-arid region, the MENA country of Algeria, has been subjected to unfavourable physical and hydro-climatic conditions accentuated by periods of chronic dryness. The climatic change which has prevailed for several decades in North Africa has touched this country in particular and has had a negative impact on its water resources.

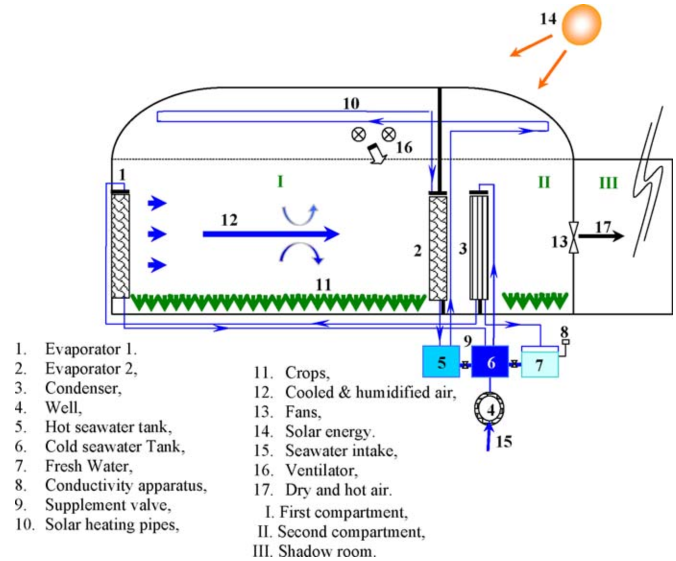


**Fig. 1.** Process schematic for Seawater Greenhouse [3,4].

The use of water for irrigation represents more than 55% of the total fresh water consumed in Algeria and is increasing. However most of the desalination research has been aimed at enhancing techniques of purifying sea water and brackish water, while neglecting improvements in agricultural irrigation efficiency [2–5,8].

Solar desalination methods are well suited for the arid and sunny regions of the world as in the Arabian Peninsula. A variety of solar desalination devices have been developed [9,10]. One of the more successful examples is the multiple effects still. Latent heat of condensation is recovered, in two or more stages (generally referred to as multi-effects), so as to increase production of distillate water and improve system efficiency. It has become apparent that a key feature in improving overall thermal efficiency is the need to gain a better understanding of the thermodynamics behind the multiple use of the latent heat of condensation within a multi-effect humidification–dehumidification solar still. In addition, while a system may be technically very efficient, it may not be economic (i.e., the cost of water production may be too high). Therefore, both efficiency and economics need to be considered when choosing a solar desalination system.

An example of a humidification–dehumidification system is a pilot plant built at Kuwait University [11]. The system consisted of a salt gradient solar pond, which was used to load the air with water vapour. Fresh water was collected by cooling the air in a dehumidifying column. In a similar study, a closed-air cycle humidification–dehumidification process was used by Al-Hallaj et al. [12] for water desalination. Paton and Davies [2] used the humidification–dehumidification method in a greenhouse-type structure for desalination and for crop growth (Fig. 1). Their Seawater Greenhouse produced fresh water and crop cultivation in one unit. It was suitable for arid regions that have seawater nearby. Abdulhaiy et al. [13] assessed the thermal performance of greenhouses with built-in solar distillation systems.

### 1.1. Process description

The sea water greenhouse uses sunlight, seawater and the atmosphere to produce fresh water and cool air, creating

temperature conditions for the cultivation of crops. The process recreates the natural hydrological cycle within a controlled environment. Seawater is pumped from a well near the sea; sand filtration keeps out solid particles and other impurities. Filtered water is send to a cold tank where it is fed in a cascade at first to the condenser then the first evaporator (Fig. 1). The brine water from the first evaporator returns to the cold tank.

The evaporator is the entire front wall of the greenhouse structure. It consists of a cardboard honeycomb lattice and faces the prevailing wind. Seawater trickles down over this lattice, cooling and humidifying the air passing through the planting area. Fans draw the air through the greenhouse and into the shade room. Air passes through a second seawater evaporator and is further humidified to saturation point. The greenhouse is equipped with a solar heating system which consists of long pipes placed along the length of the greenhouse roof. The seawater in the pipes is heated directly by sunlight's radiations and is fed into the second evaporator before returning to the hot tank. Air leaving the evaporator is nearly saturated and passes over the condenser coils carrying cool seawater, which may come from the cold tank. The fresh water condensing from the humid air is piped to storage for irrigation.

The condenser in the greenhouse desalination system constitutes the most critical component. For the greenhouse to be cost-effective the condenser has to be efficient, uncomplicated, inexpensive, and low in maintenance. A plastic condenser was built at the Al-Hail site of Sultan Qaboos University, Muscat, Oman [3–6,10]. It consisted of vertical pipes. The header and the bottom pipes were connected to two large horizontal pipes by plastic joints (Fig. 2). The water which flowed downwards was pumped through the unit; the falling cooling water discharged through a plastic pipe with its upper half cut away forming an open channel inside the assembly. The discharge pipe was inclined towards the outlet.

The Seawater Greenhouse desalination by humidification–dehumidification uses a tubular condenser with external condensation. Seawater circulates inside the tubes while vapour condenses on the external walls of the tubes. The bottleneck of the greenhouse desalination by humidification and dehumidification (SWGH) is the design of condenser. The leaks in tubes/pipes can contaminate the condensate with saline water and hence increasing the salinity of the fresh water. Details of the flow dynamics, heat and mass transfer may be resolved. It should aid in better understanding of the processes taking place in the greenhouse condenser.

The main aim of this paper was the development and comparison of mathematical models for both the current pump driven and a proposed new passive condenser in order to enhance the performance of a humidification–dehumidification Seawater Greenhouse desalination system. Passive condensation is based on the natural circulation of water, which appreciably reduces the requirements of pumps, valves and accessories. In addition to the existing recirculating pump driven system, two new designs were assessed: a passive cooling system with a condenser (IC) immersed in a water basin, and an external passive condenser (EPC) connected to a basin of water placed on top of the condenser.

## 2. Methodology

### *2.1. Experimental and simulation studies for SWGH and single condenser tube*

The Seawater Greenhouse (SWGH) was tested for the period of 18 December 2005 to 7 February 2006. The SWGH was equipped with a data logging system (Delta-T devices) which provided continuous monitoring of ambient and internal conditions; the data file contains observations at every half hour of the day. The humid air temperature and relative humidity are the two major factors affected the condensation. The relationships between the condensate rate and those variables were investigated. In this present study a mathematical model was used to determine the performances of a new immersed condenser proposed for the development of the Seawater Greenhouse. The simulated condensate values were compared with that of the actual measured values of the installed condenser.

### *2.2. Proposed designs of passive containment cooling systems*

In order to find a solution to the problems of leaks, and to optimize the condensation output of the greenhouse, it was proposed to change the existing plastic tube condenser with a passive containment cooling system (PCCS) [3]. The new system would be more economic and simpler to operate because passive condensation is based on the natural circulation of water, which reduces the requirements for pumps, valves and accessories that the current system has [1–3]. Two designs were considered: a passive cooling system consisting of a condenser immersed (IC) in water basin and an external passive condenser (EPC) connected to a basin of water placed on top of the condenser (Figs. 3 and 4).

A mathematical program was developed which can make possible the dimensioning of the new condenser, and the study of the influence of each parameter (i.e., the length and the diameter of the tube as well as the temperature of the humid air) (Table 1) on the daily fresh water production. The surface area (i.e., footprint) of the greenhouse was taken to be equal to 720 $m^2$.

The immersed condenser is similar to a traditional heat exchanger, consisting of tubes and a grill. The performance of the system is founded on the condensation of the downwards


**Fig. 2.** The condenser of the SWGH.

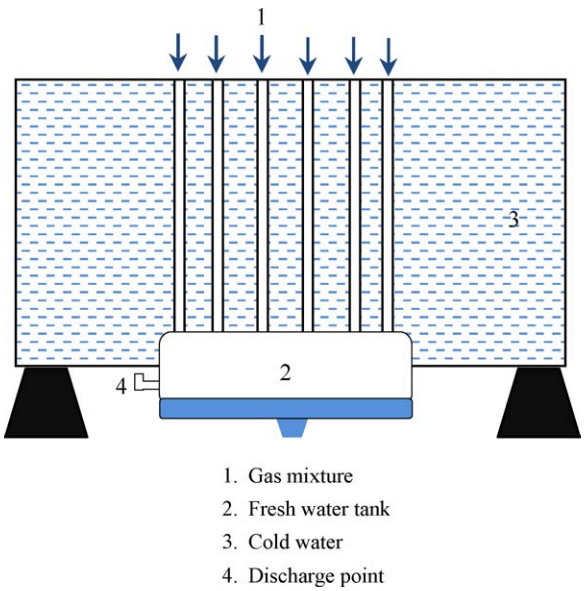


**Fig. 3.** Immersed condenser (IC).

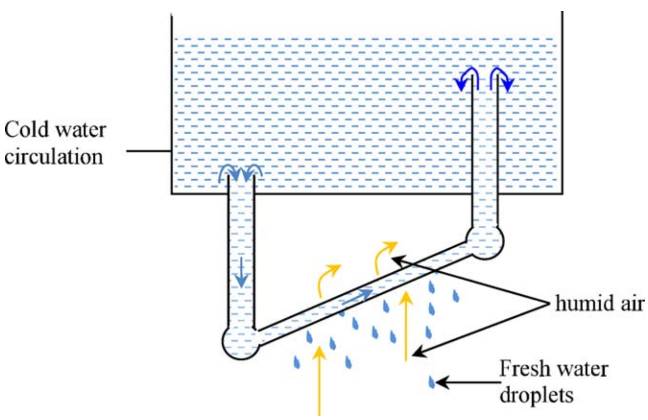


**Fig. 4.** External passive condenser (EPC).

flowing vapour inside of the submerged tubes in a basin of fresh water (Fig. 3). The condensed water on the inside wall of the tubes runs by gravitation and is stored in a tank, while the non-condensed vapour and air leave the condenser via a discharge point. This design of condenser offers flexibility in that it can be placed either inside or outside of the greenhouse seawater desalination unit. The immersed condenser (IC) as illustrated in Fig. 3 was composed of standard tubes with internal diameter and thickness equal to 0.03 m and 200 μm, respectively. Humid air at a temperature from 16 to 24 °C was assumed to circulate inside the tubes at a flow rate of 15 $m^3/s$ [14].

The second type of condenser proposed was the passive condenser with external condensation. The vapour condenses on the external surface of the condenser tubes while cooling water runs by gravity inside of the tubes (Fig. 4). This design also differed from the traditional condenser of the Seawater Greenhouse desalination unit in that no mechanical pump is required to circulate the water through the unit, similar to the IC unit.

Table 2 shows the standard composition of seawater. Sodium and chloride represent more than 85% of the total salts. However, the solubility of a solute in water depends on the temperature of the water. Fig. 5 shows the solubility of sodium chloride in water at different temperatures. From 16 to 24 °C the solubility of sodium chloride in water increases from 31.9 to 32.1 g/100 mL. It can therefore be assumed that the effect of temperature on sodium chloride solubility is negligible in the temperature interval 16–24 °C used in the present study.

**Table 1**
Main parameters of the Seawater Greenhouse [20].

| | | |
|---|---|---|
| Width | 16 m | |
| Length | 45 m | |
| Maximum height | 4.8 m | |
| First and second evaporator | 15.6 m × 2 m × 0.2 m | |
| Condenser | 15 m × 1.9 m × 0.8 m | |
| | Height of the vertical tube ($L$) | 1.8 m |
| | Diameter of the vertical tube ($D_t$) | 33 mm |
| | Total number of tubes | 4832 |

**Table 2**
Standard seawater composition [21].

| Chemical ion | Concentration (ppm, mg/kg) | Part of salinity (%) |
|---|---|---|
| Chloride, Cl | 19,345 | 55.03 |
| Sodium, Na | 10,752 | 30.59 |
| Sulfate, $SO_4$ | 2701 | 7.68 |
| Magnesium, Mg | 1295 | 3.68 |
| Calcium, Ca | 416 | 1.18 |
| Potassium, K | 390 | 1.11 |
| Bicarbonate, $HCO_3$ | 145 | 0.41 |
| Bromide, Br | 66 | 0.19 |
| Borate, $BO_3$ | 27 | 0.08 |
| Strontium, Sr | 13 | 0.04 |
| Fluoride, F | 1 | 0.003 |

### 2.3. Simulation model of the condenser

As in our case, air passes through the seawater vapour, the relative humidity is assumed to be 100%. The calculation of the mass fraction of the vapour in the gas mixture can be made using Eqs. (1) and (2).

$$\text{Mass fraction of the vapour} = \frac{\text{vapour flow rate (kg/s)}}{\text{total flow rate (air + vapour) (kg/s)}} \quad (1)$$

$$\text{Vapour flow rate} = \frac{18}{29} \times \frac{\text{vapour pression}}{\text{total pression-vapour pression}} \times \text{air flow rate} \quad (2)$$

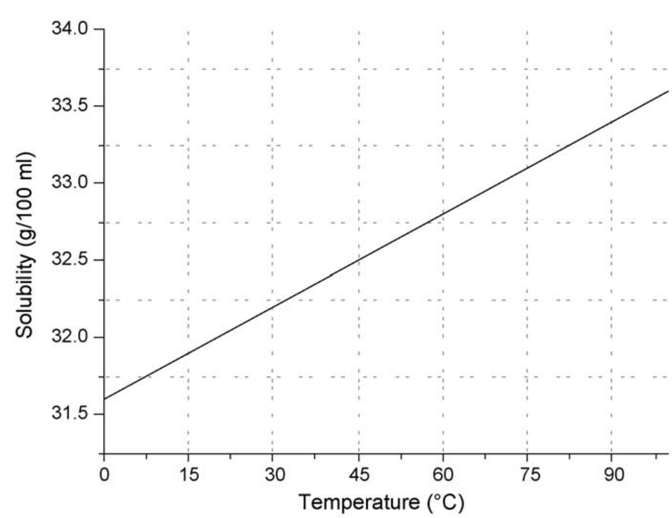


**Fig. 5.** Variation of sodium chloride solubility in water with temperature.(Adapted from El-Dessouky and Ettouney [21]).

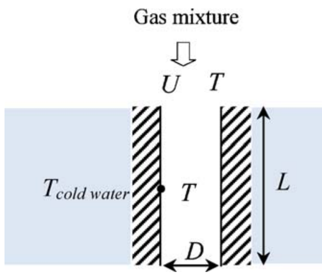


**Fig. 6.** Heat and mass transfer inside the condenser tube.

The values of the vapour mass fractions, calculated using different data for different scenarios gives values around 0.03, which is very small. In consequence, the condensation rates are quite small and the variation of the non-condensable gas fraction in the pipe is very small as long as the condensation rate is much smaller than the steam flow.

A computer model was developed to simulate the performance of the new designs for the SWG (Seawater Greenhouse) vertical condenser tube. Referring to Fig. 6 the expression of the mass transfer coefficient with the friction factor can be given by Eq. (3) [15]:

$$\frac{h_D}{U}Sc^{2/3} = \frac{f}{8} \tag{3}$$

By analogy one can write the heat transfer coefficient as:

$$\frac{h}{UC_p\rho}Pr^{2/3} = \frac{f}{8} \tag{4}$$

If the heat and mass transfer occur simultaneously, their transfer coefficients can be given by a relation resulting from the equality of Eqs. (3) and (4):

$$\frac{h}{h_D} = \rho C_p\left(\frac{Sc}{Pr}\right)^{2/3} \tag{5}$$

One can write the correlation of Dittus–Boelter for the heat transfer in the gas mixture, as follows [15]:

$$\mathrm{Nu} = \frac{hD_t}{K} = 0.023Re^{0.8}Pr^{1/3} \tag{6}$$

where the conductivity $K$ is a function of the temperature, and the Reynolds number is based on the mixture averaged proprieties:

$$Re = \frac{\rho UD_t}{\mu} \tag{7}$$

The flow of fresh water produced per unit of area can be calculated by the following relation:

$$m'' = h_D(\rho_{vap,\infty} - \rho_{vap,w}) \tag{8}$$

where [16]:

$$\begin{aligned}\rho_{vap,\infty} &= \rho_{sat}(T_\infty)\\ \rho_{vap,w} &= \rho_{sat}(T_w)\\ \rho_{sat}(T) &= \frac{2910\exp(9.48654 - 3892.7/(T - 42.6776))}{8314T}\end{aligned} \tag{9}$$

For a tube of length $L$ and a diameter $D_t$, the condensate rate can be given as:

$$\dot{m}_{con,tube} = m''(\pi D_t L) \tag{10}$$

For the vapour entering to the tube one can write:

$$\dot{m}_{vap,\infty} = \rho_{vap,\infty}UA_t \tag{11}$$

with

$$A_t = \frac{\pi D_t^2}{4}$$

In order to predict the heat exchanged in the condenser, the following parameters are needed:

- The geometry of the condenser,
- The inlet mixture temperature and pressure,
- Fraction and inlet vapour flow rate,
- The coolant temperature.

The solution sequence employed consisted of:

1. Guess a tube length $L$,
2. Guess a tube diameter $D_t$,
3. Calculate $h$ from Eq. (6),
4. Calculate $h_D$ from Eq. (5),
5. Calculate condensed mass using Eq. (10),
6. Compare the condensate rate in the tube $\dot{m}_{con,tube}$ with the condensate rate of inlet vapour $\dot{m}_{vap,\infty}$. We must find $\dot{m}_{con,tube}$ value negligible compared to $\dot{m}_{vap,\infty}$, this proves that one can ignore the change in air mass fraction in the tube,
7. Go back to step 1 until the above iteration converges.

$$\frac{\dot{m}_{cond,tube}}{\dot{m}_{vap,\infty}} < 10^{-3}$$

The maximum height of the Seawater Greenhouse front was 1.9 m, technically; the condenser tubes lengths must be less than 1.9 m. We proposed three scenarios (i.e., tube length equal to 1.8, 1.2 and 0.6 m).

## 3. Results and discussion

### 3.1. Experimental and simulation studies for SWGH and single condenser tube

According to a recent research done by Mahmoudi et al. [3] based mainly on a statistical analysis of the collected weather data, it was found that the greenhouse produces 98% of total freshwater in only 8 h (between 09:00 and 17:00 h). Fig. 7 shows the comparison between the measured fresh water produced by the Seawater Greenhouse and the relative humidity of the air mixture entering the condenser. Looking at Fig. 7, according to the hours of the day of 9 January 2006 it can be seen that Seawater Greenhouse begins to produce fresh water when the relative humidity in the space

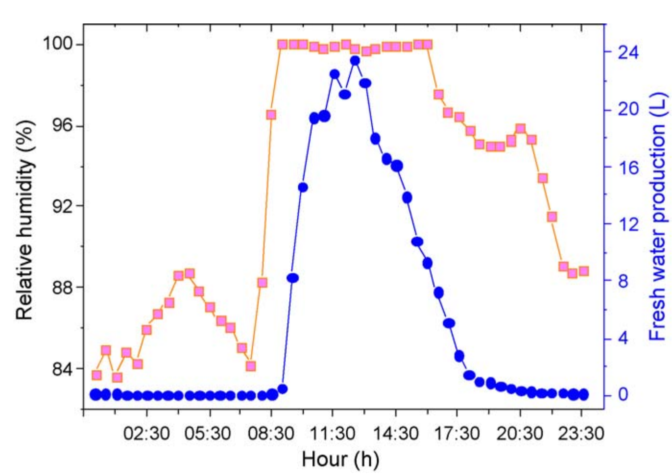


**Fig. 7.** Comparison between diurnal relative humidity and the measured fresh water produced by the Seawater Greenhouse (SWGH) [3].

between Evaporator 2 and the Condenser (see Fig. 1) approaches 100%, and decreases when relative humidity decreases. Note that humid air enters Evaporator 2 and leaves the unit near saturation.

It can also be seen that the relative humidity values were high (≈100%) between 08:00 and 14:00 while lower values (<90%) were seen between 14:00 and 21:00 h. It should be noted that in the interval extended from 09:00 to 17:00 h, the Seawater Greenhouse produced 98% of the total daily freshwater. This is consistent with the interval when the relative humidity values were high. For these reasons, our mathematical model supposed humid air entering the condenser was around 100% relative humidity.

Fig. 8a and b show the variation of the measured and predicted mass condensate rates in a single tube, respectively, of the installed condenser and the proposed immersed condenser. The experimental data were depicted every half an hour. It can be noted that the two plots of the measured and the predicted mass condensate rates have the same trend. The results suggest that the mass condensate rate of the immersed condenser's tube is significantly higher than the actual installed condenser's tube.

In order to enhance the condensation output of the greenhouse, it was proposed to change the existing plastic tube condenser with a passive containment cooling system (PCCS) [3]. The new system reduces the requirements for pumps, valves and accessories that the current system has [1–6]. Two designs were considered: a passive cooling system consisting of a condenser immersed (IC) in

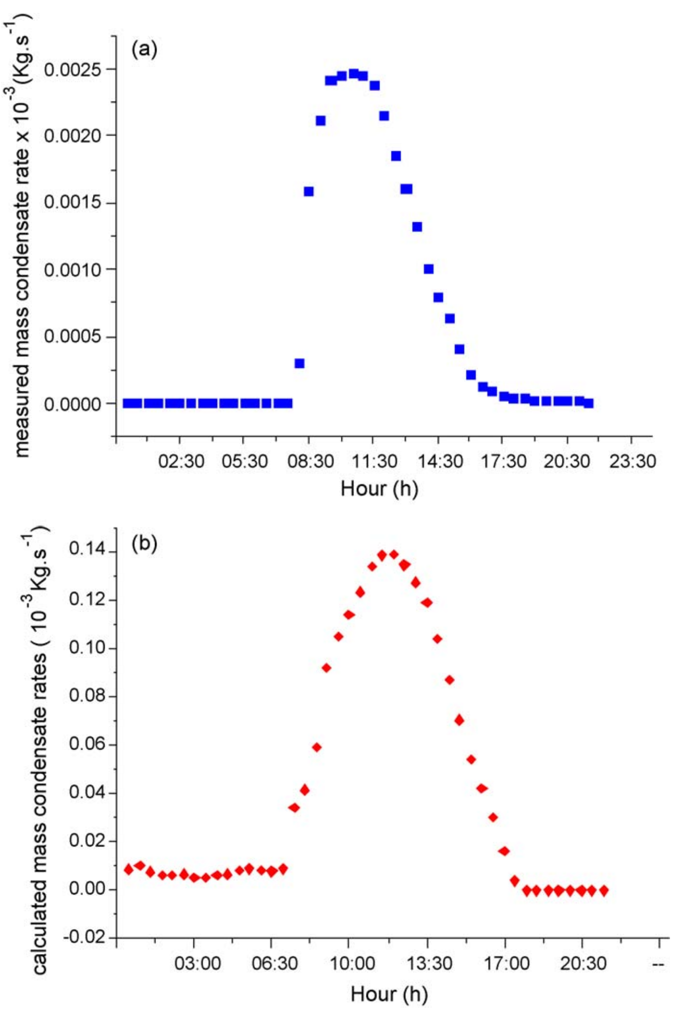


**Fig. 8.** (a) Diurnal measured mass condensate rates of one installed condenser's tube in the Seawater Greenhouse (SWGH). (b) Diurnal calculated mass condensate rates of one proposed immersed condenser's tube in the Seawater Greenhouse (SWGH).

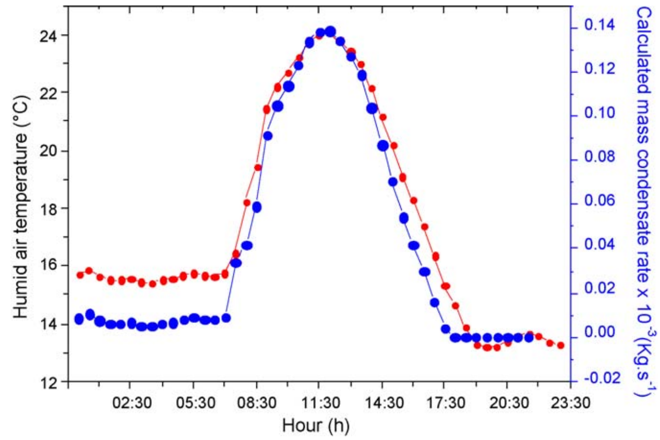


**Fig. 9.** Comparison of the diurnal calculated mass condensate rate of the immersed condenser's tube and the humid air temperature.

water basin and an external passive condenser (EPC) connected to a basin of water placed on top of the condenser (Figs. 3 and 4).

### 3.2. Simulation study of proposed immersed passive condenser system

Fig. 9 depicts the variation of the diurnal calculated mass condensate rate of the immersed condenser's tube and the humid air temperature which drawn every half an hour. The two plots were seen to follow the same trend (i.e., condensate went hand-in-hand with humid air temperature).

Fig. 10 shows the variation of fresh water production with the cooling water temperature and tube length, it can be concluded that for a given cooling water temperature, the rate of condensation increases with increasing length of the tube. An economic study can define a compromise between the length of the tube to be used and the necessary temperature of the cooling water.

Fig. 11 illustrates the variation of the condensation rates with the tube inner diameter for the three scenarios. High condensation rates were found for large diameters and low cooling water temperatures.

### 3.3. Economic considerations

In the present study with the new condenser designs it was shown that the Seawater Greenhouse produce 40.25 $m^3$/day of

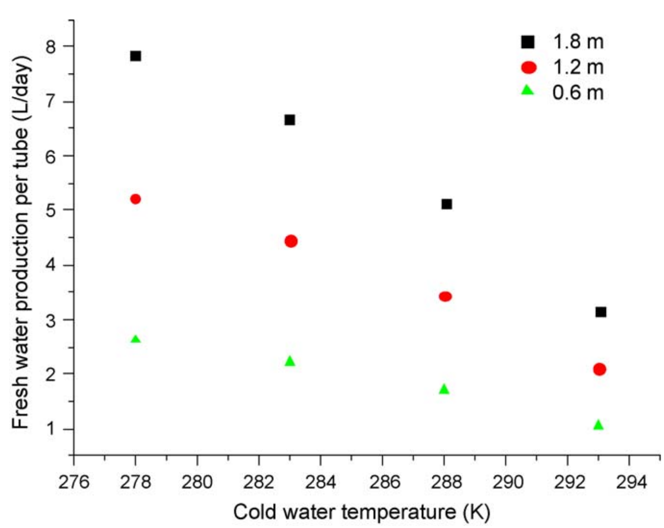


**Fig. 10.** The variation of fresh water production with the cooling water temperature for three different condenser tube lengths (i.e., 1.8, 1.2 and 0.6 m).

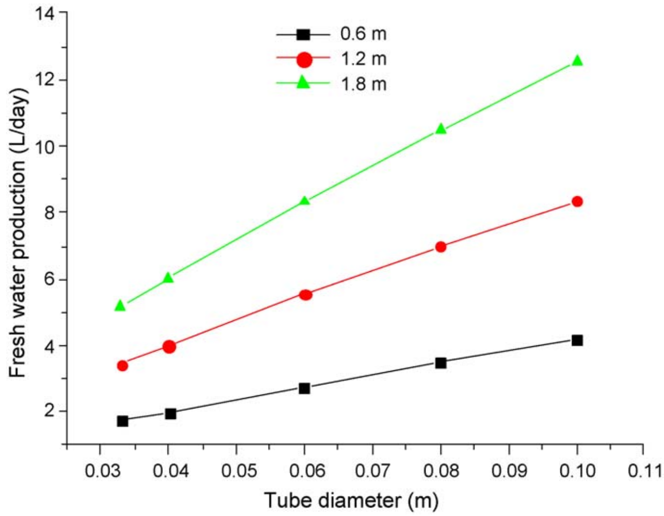


**Fig. 11.** The influence of the condenser tube diameter on the fresh water production for different tube lengths.

fresh water compared to 0.298 $m^3$/day produced by the actual installed condenser. Due to the elimination of the recirculation pumps, the power consumption will certainly decrease significantly. In a similar simulation study by Sablani et al. [8]. on the Seawater Greenhouse, they examined the total fresh water production for three climate scenarios as well as testing the performance sensitivity of their model for various greenhouse designs. Their model results predicted that the Seawater Greenhouse would perform efficiently throughout the year, but with measurable variations in performance between the alternative versions. For example, the water production rate and energy efficiency results from the simulations using optimized and constant values for fan and pump speeds showed that a temperate scenario had almost double the water production rate per hectare compared to a tropical scenario (i.e., 20,370 $m^3$/ha compared to 11,574 $m^3$/ha) while the power consumption for the former was only slightly higher (i.e., 1.9 and 1.6 kW h/$m^3$, respectively).

In the case of SWGH condensers with pumps, Sablani et al. [8] showed that water productivity can be improved but with greater energy consumption, and that efficiency can be improved but with a small reduction in water output. We can speculate that it may be possible to overcome these problems with the proposed passive condenser systems, since they do not require mechanical pumps.

Cost estimation of solar desalination systems is a complicated process. Delyannis and Belessiotis [17] noted that solar energy conversion plants are capital-intensive enterprises. Hoffman [18] presented a detailed theoretical cost analysis based on operational data for solar driven desalination plants. Voivontas et al. [19] analyzed water management strategies based on desalination systems powered by renewable energy sources such as wind and solar energy. The cost of alternative solutions, taking into account energy costs or profits by selling to a grid, was estimated. The present study suggests that capital and operating costs can be reduced by using a passive condenser system which eliminates pumps and valves. This needs to be verified experimentally. Furthermore the addition of renewable energy systems such as solar panels and wind towers may also help to reduce operating costs.

## 4. Concluding remarks

In the present study a mathematical model for a proposed new immersed condenser was developed and assessed in order to enhance the performance of a humidification–dehumidification Seawater Greenhouse desalination system. Preliminary results suggest that the passive condenser has a much greater water production capacity than the existing pump driven system. While the model for the proposed system still needs to be validated experimentally the initial study indicates that the passive containment cooling system is a promising improvement in the further development of greenhouse desalination.

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